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ACCELERATED LIFE-TIME TESTING OF BAe-STIRLING CRYOCOOLER WITH LINEAR DRIVE
PART I; FATIGUE RELATED TESTING
(Final report)

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Kharkov -1999

AQF00-02-0420

1. INTRODUCTION TO THE PROPOSED ACCELERATED TEST PROCESS

Modern space cryocoolers should have an expected lifetime of 10 to 20 years. As a result, it is virtually impossible to perform a meaningful real time test of a design or of actual flight hardware before the actual system becomes obsolete. Until recently a method for a realistic accelerated did not exist. The primary reason for this has been that it was essentially impossible to generate a simulation of the accelerated external environment and prediction models for failure of various cryocoolers elements was poor.

Based on 30 years of experience aT the Special Research and Development Bureau (SRDB) of the Institute for Low Temperature Physics and Engineering (ILTPE) in Kharkov, Ukraine in the field of development and tests of space cryocoolers we have developed and have validated a new, innovative approach to accelerated test. Our test methods allow us to verify the lifetime design of a predicted 10-year system in less than one year.

According to our approach all critical elements of a Stirling type cryocooler with a linear drive (see fig. 1) can be placed into two basic failure modes; time (aging) and fatigue. For elements which normally fail due to aging it is possible to simulate the failures by the accelerated change of vacuum of a cryostat in which the cooled object and displacer cooled conductor are placed. The degree the working substance (helium), looses its regeneration capability can be related to blocking up of the regenerator by cryodeposit from outgassing products etc.

Areas where fatigue failures are most likely to occur are the spring suspension of pistons, windings and flexible connecting wires of drives of the compressor and displacer flexible and pipelines attached to elements. The tightness of the cryostat and cryocooler body could also be expected to fail from fatigue. The test methodology for accelerated fatigue related elements is by periodic mechanical loading with a cyclical change of temperature, especially with non-uniform fields of temperature.

The basic approach to accelerated test consists of the definition of a complete set of critical cryocooler elements and the parallel individual accelerated tests using unique techniques. The system test of the structurally connected group of elements is performed as well. As we increased the frequency during

test we simultaneously maintain all the normal functionality of other parameters. The cryocooler tests are assumed successful if all its critical elements have maintain their functionality during the period of test.

In order to allocate classes of critical elements a validated methodology is used. The duration of actual test for these elements can be reduced 10 to 20 times using a projection method which takes approximately 1 to 2 months.

For critical elements of the time related class the limiting definition of critical parameters is determined aT the moment of cryocooler failure, i.e., cryodeposit mass blocking of the displacer and the average rate of change of the parameter during its lifetime (for example, cryocooler outgassing rate and containment). For wear of the gap sealing the allowable gap wear and rate of wearing typify the limiting parameters.

For critical elements with fatigue type of aging we propose to operate the cryocooler aT an increased frequency of 4 to 5 times the normal frequency of operation. To approach such frequency for membrane suspension of the compressor piston and the total drive (winding, piston, magnets, wires) the helium removal from the compressor helps. It results in the elimination of work related to the compression of gas and the associated friction. It also decreases by a factor of 10 the current of the drive feeding while maintaining its operating temperature.

For the accelerated tests of the compressor body related to tightness as it experiences thermal cycles aT an increased frequency by approximately one order of magnitude of heating and cooling in combination with widening of the range of temperature cycles from a range of operation of -35^0 C to $+75^0$ C to $\pm75^0$ C.

This proposed proven approach allows us to realize a resultant equivalent system accelerated cryocooler test.

The test process in general will consist of the following: (1) Test the cryocooler for a "zero" operating time with exposure to a defined complete set of characteristics (temperature of the cold finger, cooling capacity, consumption capacity, level of vibration, etc.) aT the normal operating frequency. (2) Following this we perform individual accelerated tests of the critical elements with identical characteristics of acceleration for each element. The elements will experience an operating time equivalent to 1,2, 3, 4

and up to 10 years of normal operation. The repeated testing of cryocooler characteristics aT its working frequency is made for each of the specified intervals. After the performance of tests which have are equivalent to "10 years" of accelerated operating time we are able to establish if the total cryocooler characteristics meet the allowable limits over time.

The offered technique of the accelerated tests requires insignificant modifications to the integrity of the test artifact for fatigue test. The modifications consist of connecting two additional branch pipes to the compressor body for the purpose of removing, filling and blowing through of the compressor gaseous helium.

In the subsequent units of the report the descriptions of designs of stands for the accelerated tests and substantiation of offered techniques of the accelerated tests are given. For this effort we are focusing our attention to accelerated fatigue testing. The follow-on effort under new contract will address in detail the accelerated lifetime testing.

2. Description of Fatigue Accelerated Test Stands and Test Operations

In accordance with the SRDB European Office for Aerospace R&D (EOARD) and Orbita Ltd. contracts three (3) stands have been developed for accelerated fatigue test. Figure 2.1 illustrates the design of each.

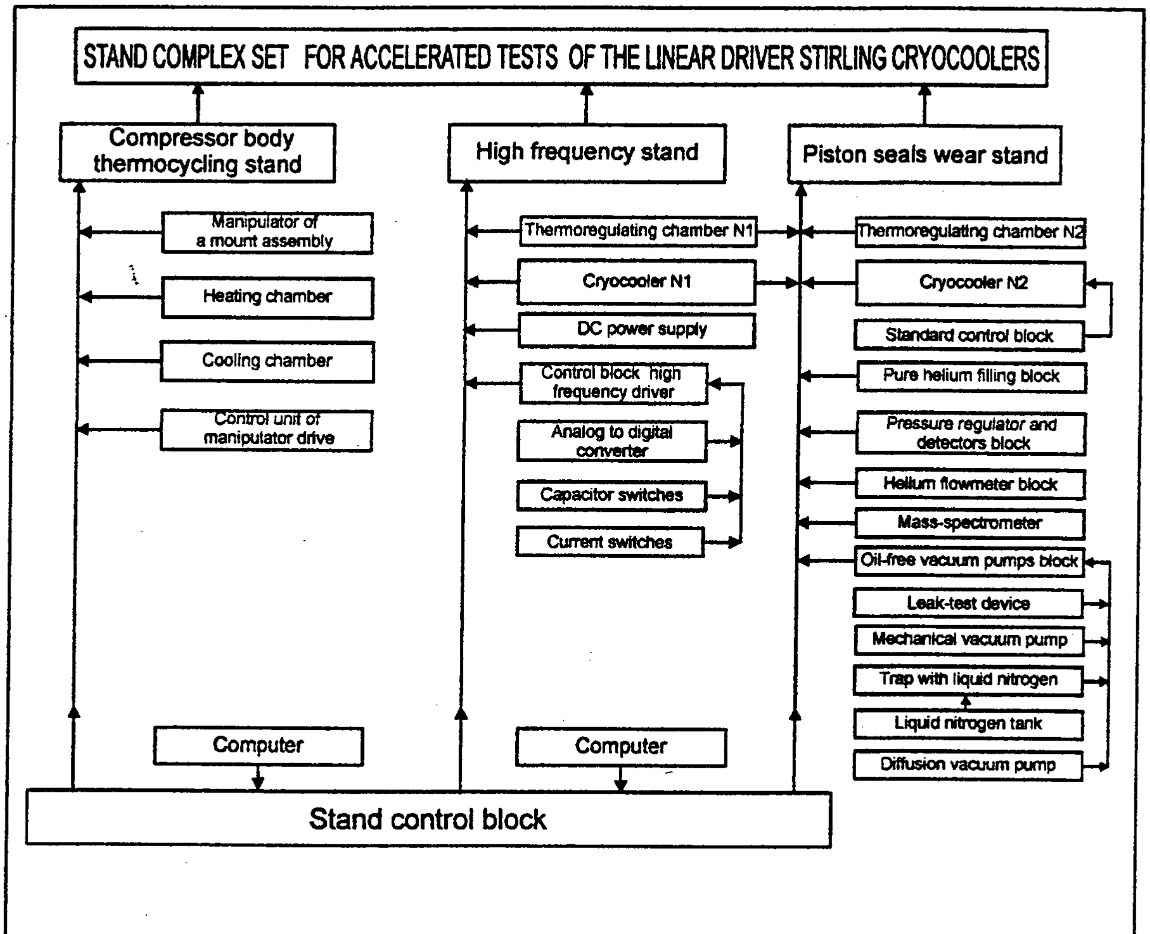

Figure 2.1

1. The stand for a accelerated testing of compressor linear drive.
2. The stand for control of the compressor gap sealing wear during accelerated testing.
3. The stand for accelerated testing of the compressor body related to permeability after thermal cycling.

Stand 1 is for simultaneous testing for a period of 1.5 to 2 years duration of two compressors. The first compressor is tested aT the normal operating frequency with the aid of standard Customer test equipment. Compressor 1 during this test is fixed inside a thermal regulating chamber which maintains the temperature aT +40°C. This compressor is also connected to stand 2 for the control of gap sealing wear.

The second compressor is tested aT 4 to 5 times higher frequency than the first one. It also is connected with a block attached to the compressor aT the high frequency and with the stand for gap sealing control.

The second compressor is also located inside a thermal cycling chamber which is maintained in the temperature range -20°C to -40° C.

Prior to beginning and after ending the high frequency testing of compressor 2 its qualification testing aT standard frequency can continue aT the discretion of the Customer. Before testing on stands 1 and 2 the both compressors undergo minor modification (see Table 2. 1.). Two tubes are connected to each of them. Through each helium is filled, evacuated or blown through the gap sealing.

Compressor 2 is placed on stand 3 after it completes after it completes the high frequency and standard frequency tests. Prior to this test compressor 2 is disconnected from stands 1 and 2 and filled with helium aT the operating pressure and sealed. On stand 3 compressor 2 undergoes thermal cycling within the temperature range +75° to -75°C. After the thermal cycling is completed the level of helium leakage from compressor 2 body is measured.

How each of the two cryocoolers are modified for accelerated test is shown in Table 2. 1. The compressor 1 and 2 complete testing plan is shown in Table 2. 2. ; 2. 3.

Table 2.1.

Cryocoolers modernization for accelerated testing.

Cryocooler type	**Modernization type**	
	Connection of 2 evacuation tubes	**Additional position sensors of piston (if possible)**
Cryocooler 1 for standard frequency	+	-
Cryocooler 2 for accelerated testing	+	+

The testing plan of #1 cryocooler with standard frequency operation is presented in Table 2.2.

The testing plan of #2 cryocooler with accelerated frequency operation is presented in Table 2.3.

Table 2. 2.

#1 Cryocooler testing schedule for standard frequency

Row #	Task number	Months 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
1	Cryocooler parameter standard testing of primary parameters (1year)																		
2	Helium evacuation																		
3	Helium blowing through compressor seals to determine gas permeability																		
4	Helium 1 day recirculation through cryocooler and cryosorbing pump to determine cryocooler outgassing																		
5	Helium backfill																		
6	Cryocooler and compressor thermal cycling aT 10 times normal and the helium permeability determination																		

Table 2. 3.

#2 Cryocooler accelerated testing schedule

Row #	Task number	Months 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
1	Cryocooler main parameter testing aT standard frequency																		
2	Cryocooler assembly with test stands																		
3	Helium evacuation																		
4	Accelerated testing of compressor aT frequency about 400 - 500 Hz																		
5	Helium blowing through compressor seals to determine gas permeability																		
6	Accelerated filling of cryocooler body and cryostat representing 10 years outgassing products (excluding steam)																		
7	Cryocooler displacer cooling																		
8	Accelerated cryocooler filling with steam																		
9	Helium backfill																		

3. PRINCIPLES OF THE OPERATION AND STRUCTURE OF STANDS.

3.1. STAND FOR THE ACCELERATED TESTS OF AN ELECTRICAL LINEAR DRIVE COMPRESSOR.

3.1.1. The Basis for Increasing Frequency.

Condition for complete modeling of compressor drive tests on high frequencies are represented by the following equation [1]:

dm = idem, I = idem, F = idem, T = idem,

idem - identical test parameters with the standard frequency unit, amplitude of the piston movement - dm, the winding current - I, the moving force - F and temperature, T of the total system. Furthermore, we should delete the requirements I = idem and F = idem.

Based on the conditions generated in the standard case and the helium removal from the compressor, the maximum achievable frequency of tests will be defined by the equation:

$$f_{acc} = \frac{1}{2 \times \pi}\left[\frac{F}{d_m \times m}\right]^{\frac{1}{2}}$$

Where facc - maximal linear frequency of the accelerated test and m - mass of the moving elements of a drive.

Typical for the test cryocooler the following parameters are anticipated: F = 20 N, dm=$4*10^{-3}$ m, m = 10g and the nominal frequency equals 35 - 50 Hz. From the above equation the maximal frequency of the drive movement with a supply power of Psup = 25 W is 113 Hz. This is 2.25 times nominal operating frequency 50 Hz of a compressor operating using gaseous helium as the working fluid.

In addition, during the removal of helium there is practically one more opportunity to increase the frequency by increasing the electrical power. This is based on the relationship for the windings, P0 = I^2R where R - resistance. If the power can be raised to aT least the level of the nominal power of the drive - 25 W. In this case, the current in the windings and the force of the drive will increase in 3.16 times, and the frequency of tests facc will be raised

approximately $\sqrt{3.16}$ = 1.78 times. This result represents a frequency for the tests to be aT a level approaching 200 Hz. This actually exceeds the nominal frequency by a factor of 4.

Such an increase of the frequency will be accompanied by an increase of resistive losses Re by an order of magnitude on the drive windings. Nevertheless such an increase P0 is possible practically without infringement on the operating temperature regime of the drive and compressor. The basis for this assumption is that the basic energy of the drive (about 90 %) is spent on compression and friction of helium. Therefore helium removal from the compressor has permitted us to effectively reduce the power by an order of magnitude. The energy is consumed by the drive, resulting in the level of resistive losses, $P0 = I^2R$.

After helium removal the winding resistance aT the high frequency is kept constant based on the condition, I = idem. This allows an increase of the current 3.16 times while keeping the heat emission with a constant nominal power of 25 W. This means the thermal regime of the windings, even with the increased frequency will remain constant.

One more factor which contributes to the destabilization of the thermal regime is the overheating of the membrane suspension of the drive due to the rise in internal friction of the material of suspension with increased frequencies. The estimations show, that with an order of magnitude increase in frequency the overheating of suspension can not exceed 50° C. To preclude suspension overheating, we perform the accelerated tests taking the two following measures.

1 - Helium from the compressor is not completely evacuated, but its pressure is reduced to 1 atmosphere level. It allows us to maintain a high level of heat conduction from the winding suspension to the helium, and further to the compressor body through a gap between the piston and cylinder.

2 - During high-frequency tests the compressor is located in the thermal regulating chamber which is support by its attach points. The temperature is maintained aT -40° C. This results in an improvement of the heat conductivity from the suspensions to the compressor body through the zone of fastening of suspensions in the body. For the same reason we do the same for the second compressor. With the help of the second thermal regulating chamber we maintain maximum test temperature equal to + 50° T with a nominal operating frequency.

The reduction of kacc factor (kacc is the element of the frequency which increases aT high test frequency) results in an increase of frequency up to a factor of 4 instead of the previously stated 6 - 10 times. This is a result of the absence of any compressor modifications. These modifications would have been the introduction in a winding of a bypass coil and the removal of the piston from an axis of the drive.

We have considered one more opportunity to increase the frequency, facc, by the secondary increase of a current I and force F a factor of 1.56. This gives us the possibility to not exceed the nominal power of the drive more than 2.5 times. Then the frequency of the accelerated tests will be once again raised $\sqrt{1.56}$ = 1.25 times and will achieve a level 250 Hz. Therefore, the increase of frequency will achieve a level 5 times. However, it will require significant complication of the test circuits. The increase of the current will be a factor of 2.44 (proportionally I^2R) with an associated resistance loss of 1.56 times on the drives and will achieve a power level 61 W.

There is one additional option to keep a temperature regime of the compressor drive aT room temperature. We can introduce some helium flow from within the refrigerator, itself, as a coolant for the compressor drive. Its introduction can be carried out through a pipeline connected to the compressor with an additional pipeline as a return. This approach is technically very difficult and consequently aT this stage we have not considered practical ways of implementing it.

3.1.2. Description of Stand Operation.

The stand for accelerated tests of the compressor electric drive must be capable of allowing the drive to operate aT significantly higher frequencies than the standard regime. The stand construction is caused by three main factors:

1. Inductive resistance of a winding is essentially increased with frequency growth. This is due to its decrease current output. In order to store the prior winding power supply it is necessary to increase the drive voltage output. Therefore, it is necessary to use the phenomenon of an electromechanical resonance (EMR) in the electric drive (ED) to compensate for the influence of inductance.
2. With growth of frequency it is essential (by square-law dependence) to increase the supply power. This is necessary for maintenance of the appropriate acceleration. In order to

reduce drive power significantly lowered the compression work of compressor gas. However, in order to accomplish this it was necessary to introduce a round- about method for connecting the compressor output with the carter.

3. Due to increase in frequency the quantity of heat emission sharply increases (proportionally with the fourth degree of frequency) in a drive winding. As a consequence, the winding temperature grows. For this reason a stand power electronic block must be constructed for maintenance of the necessary winding current and the pushing force in order to achieve frequencies of 500 Hz. However, the requirement for removing allocated heat (for reduction of the thermal mode of the drive elements) necessitates the application of a winding cooling system. As a result, the efficiency of this heat removal process we maximize the magnitude of the drive operation frequency.

The optional methods for increasing the operational frequency of the test compressor are determined first of all by mass of moving parts and design of the drive winding. The stand structural construction is shown on fig. 3.1.1. Commands from the personal computer (PC) (1) through the keyboard, monitor and the adapter (2) (peripheral devices) (3) enter to the controller (4). In the controller they will be transformed to electrical signals specifying a mode for managing units. The frequency installation program through control frequency device (5) provides a command, going from the block of managing of the switchboard of capacitors (BMSC) (6) for connection of a selected capacitor in a circuit of ED winding with help of the capacitors switchboard (7). The pulse shaper (8) provides generation of pulses of the required frequency that appropriates EMR in the drive with the given capacitor. The generated pulses operated by the four- arm power switchboard provides of course 10 pulses of a reversal current through a drive winding. The pulse current size is established by the block of the force current (11) by the signal of the current control device (12). The managing signal for control device (12) in the controller (4) is developed under the PC software providing the required amplitude of a drive movement. The power supply block (13) provides the fixed value voltage which is maintained aT the maximum demand value for the winding current. Within the data units (14) connected with the winding or mobile drive element there are the axial position (DUAP) measurements, winding temperature measurements, compressor body temperature, etc. The analog-digital converter (ADC) (15) provides processing of signal data for transfer to the controller (4) and PC (1).

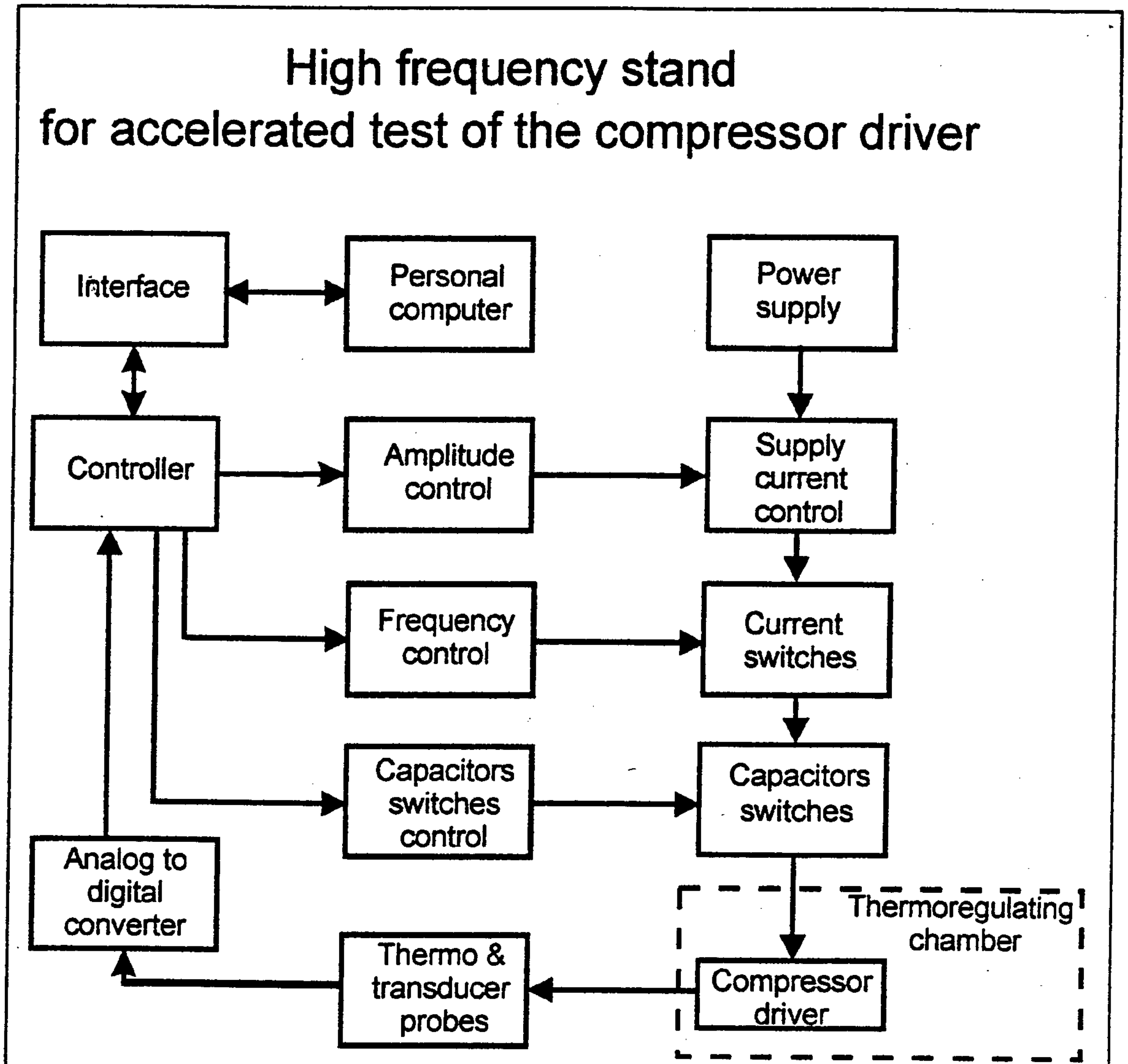

Figure 3.1.1 Stand Structure

The thermal regulating chamber provides demanded temperature of the compressor body.

The general view of the stand is given on fig. 3.1.2.

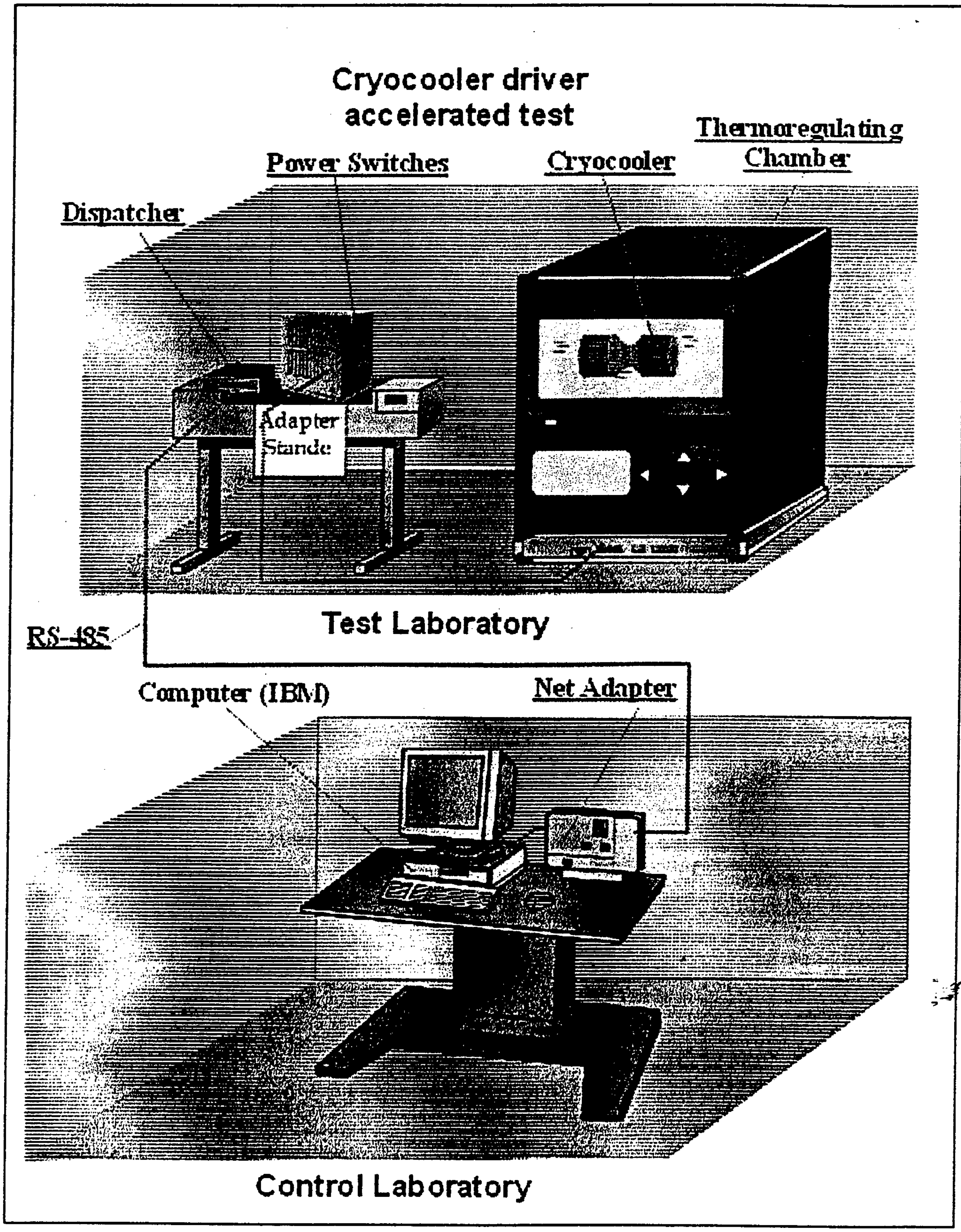
Cryocooler driver
accelerated test
Power Switches
Cryocooler
Thermoregulating
Chamber
Dispatcher
Adapter
Stande
Test Laboratory
RS-485
Computer (IBM)
Net Adapter
Control Laboratory

Figure 3.1.2

3. 1. 3. Computation of thermal properties of the drive winding.

The following formulas are applied for realization of drive thermal calculations:

$$x = \frac{1}{2} a \left[\frac{\tau}{4} \right]^2;$$

$$a = \omega^2 \cdot x;$$

$$w = 2\pi \cdot f;$$

$$F = m \cdot a;$$

$$F = B \cdot l \cdot I;$$

$$V_{\max} = a \cdot \frac{\tau}{4};$$

$$E_k = \frac{m \cdot \left[V_{\max} \right]^2}{2};$$

$$A = F \cdot x = E_k;$$

$$W = I^2 \cdot R;$$

where *a* - maximum possible acceleration in the drive, *m/s²*;

f- cyclic frequency, *Hz*;

ω- circular frequency, *rad/ s*;

x- amplitude of fluctuations, *m*;

Vmax- maximal speed in the drive, *m/ s*;

τ- period of fluctuations, *s*;

m- mass of the drive mobile part, *kg*;

Ek- maximum possible kinetic energy of mobile elements;

A - work;

F- maximum value of efforts in the drive, *N*;

B- induction of a magnetic field in the working backlash, Ò;

l- length of a drive winding wire the backlash, *m*;

I- peak value of a current in the drive winding, *A*;

W- heat emission capacity in a drive winding, *W*;

R-resistance of the drive winding for a constant current, Ω.

The estimated values are executed for a drive with the following parameters:

m = 0.01 *kg*, õ = 0.004 *m*, Â = 0.8 *T*, l = 9 *m*, R = 1 Ω.

The work of gas compression in the drive is excluded by the introduction of by-pass tube.

The data values of current amplitude in drive windings and thermal capacity, allocated in a winding, are given in table 3.1.1.

The most significant contributor for heat emission is the mass, m. For an illustration in the table the results of computations for two values of mass (10 and 50 g) are given. In this option the resultant resistance of a wire winding R is equal to 0.329 Ω.

By the requirements for cooling efficiency of the drive winding there is an accepted range of excess temperature of the winding over environmental temperature. In the determination of the heat transfer process we must consider the transfer from the winding to the environment, the heat transfer from the winding to the compressor gas, the heat transfer from gas to the compressor body and heat transfer from the body to an environmental air.

Temperature of the winding is determined by the following dependence:

$$T_w = T_e + dT_1 + dT_2 + dT_3$$

Tw - temperature of the winding,

Te - temperature of the environment,

dT1 - temperature difference between the compressor body and environment,

dT2 - temperature difference between helium in the compressor and the compressor body,

dT3 - temperature difference between the winding and helium in the compressor.

For the computations it is accepted that:

- The compressor body with mass of mobile parts 10 g represents the cylinder with a diameter of 100 mm, area of the heat exchange surface $2.84 \cdot 10^{-2}$ m^2.
- The compressor body with mass of mobile parts 50g represents the cylinder by a diameter of 100 mm, area of the heat exchange surface $8.33 * 10^{-2}$ m^2.

- For both variants it is accepted that outside compressor is blown through by room temperature air in a direction perpendicular to the axis of the cylinder with operating speed of 10m/c.
- The compressor is filled by helium with pressure of 0.12 Mpa and the heat exchanger surface area is equal to the outside surface area;
- The heat exchange of the drive winding located in the magnetic system working backlash is considered as heat exchange in the flat gap in which both surface contribute to the heat exchange. It is also accepted that the gap is equal to 0.5 mm on the each side, speed of the gas movement in the gap is equal to the average speed of the piston the heat exchanger surface area is equal $7.147 \cdot 10^{-3}$ m^2.

The results of temperatures difference computations for various frequency fluctuations and corresponding current and heat exchange power in a winding values are given in tab. 3. 1. 1. and are shown on fig. 3.1.3.

Table 3.1.1.

Parameter	**Mass, g**	**Frequency, Hz**					
		50	**100**	**150**	**200**	**250**	**300**
I, A	10	0.73	214	4.43	7.6	11.6	16.6
	50	2.75	9.75	21.3	37.3		
W, W	10	0.794	6.75	28.9	85.1	200.0	405.1
	50	2.48	31.3	148.5	457.3		
dT1,^{0}C	10	1.49	7.62	27.3	74.8	170.1	338.7
	50	2.15	27.1	129	397.1		
dT2,^{0}C	10	2.36	7.03	18.9	44.9	98.8	207.8
	50	3.45	26.1	106.4	356.7		
dT3,^{0}C	10	0.34	2.8	11.1	28.4	54.6	87.2
	50	0.295	3.34	12.1	24.8		
The sum.	10	4.18	17.44	57.3	148.1	323.6	634
^{0}C	50	5.9	56.7	247.5	778.6		

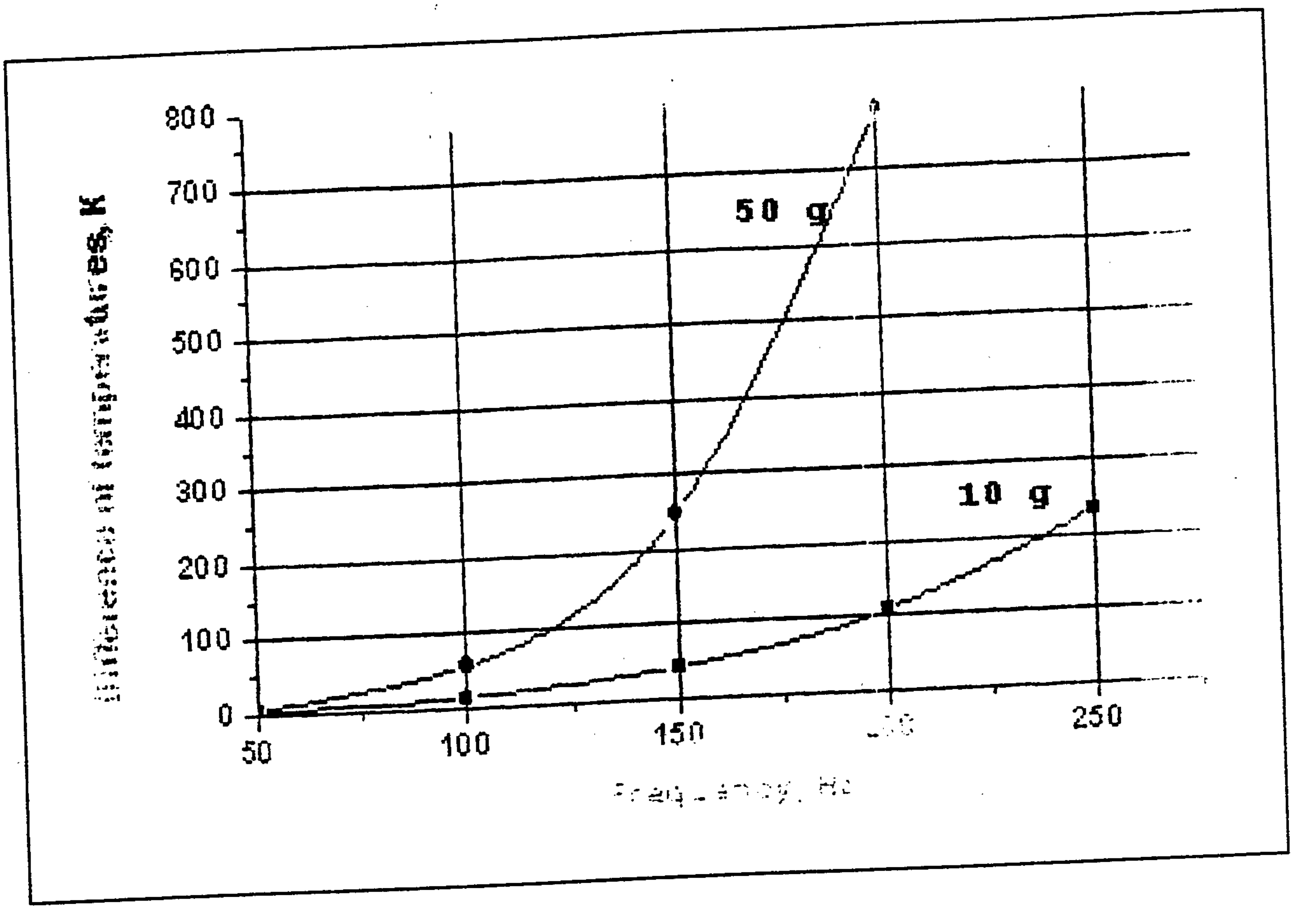

Figure 3.1.3. Temperature versus Frequency for
m = 10 g and m = 50 g.

Considering the above results the following can be concluded:

- The achievable frequency may be determined by knowledge of the heat transfer.
- If the input temperature equals the temperature of the drive winding and this equals 100°C and the test compressor is placed in a chamber maintaining 25°C temperature, than the actual achievable frequencies will be 185 and 120 Hz for compressors with a mass of the moving systems being 10 g and 50 g respectively.
- If the compressor is placed in a thermal regulating chamber maintaining a temperature of -40°C and fluid flow rate is 10 m/s we can achieve frequencies of 225 Hz and 130 Hz for compressors with masses of moving parts being 10g and 50g respectively.

- If the compressor is placed in a thermal regulating chamber maintaining a temperature of -40^{0}C and fluid flow rate is 10 m/s we can achieve frequencies of 225 Hz and 130 Hz for compressors with masses of moving parts being 10g and 50g respectively.
- If we blow helium through the compressor carter to complete the compressor cooling circuit, then as a result of the removal of heat losses between the environment and the compressor body and also between the compressor body and compressor gas, it is possible to increase working frequency.
- Temperature of a drive winding is defined by the following dependence:

$$T_o = T_k + dT_3 ;$$

where Tt - temperature in the thermal regulating chamber.

•The characteristics of the frequency behavior of the compressor temperature change of the drive winding curve shows that the frequency can easily be increased to a level of 300 Hz for a moving mass of 10 g and for compressors with a moving mass of 50 g it is possible to increase the frequency to 150- 160 Hz

The stand's basic characteristics are presented in a table 3.1.2.

Table 3.1.2.

Row #	Parameter	Parameter value
1	Frequency range, in presence of the electric-mechanical resonance for several frequency values, Hz	30-300
2	Power capacity of one channel, W	500
3	Power voltage, V, DC	27±3
4	Current impulse form: half period time or less - one direction current, other half or less - back direction current	
5	Drive winding current maximally, A	
6	Life- time, years	3
7	Stand cooling type	3
8	Electrical parameters of drive windings:	by air
	Resistance, ohm	0.5- 1.0
	Inductivity, mH	0.5- 1
9	Maximal frequency (Hz), aT room temperature compressor with moving mass: 10 g 50 g	185 120
	Maximal frequency (Hz), for compressor in thermal regulating chamber with moving mass:	

	50 g	225 130
10	Displacement amplitude maximum, mm	±4
11	Stand must have control computer control unit*	
12	Position control of moving parts by position detector	
13	Thermal regulating chamber* for tested compressor	

3.2. STAND FOR THE CONTROL OF THE WEAR OF THE COMPRESSOR GAP SEAL

3.2.1. The basis for the methodology of gap seal wear projection.

There are three possible reasons that we have identified that can cause a dimensional change of the gap seal for the compressor piston, resulting in a helium leak through the gap.

1.The presence of radial deformation of the membrane suspension results from the pressure induced by radial forces. Such forces can appear as a result of inaccurate assembly of the friction sensitive unit as well as the appearance of radial oscillations of the membrane suspension during the compressor operation aT a standard frequency. The leakage increase due to the radial deformation can be as much as 2.5 times. These deformation differences are the result of piston position change from an axis-symmetrical one (along cylinder axis) to a parallel displacement to cylinder generatrix one and piston contact with it. A relation of gap gas conductivity for these two edge positions can also approach a multiple of 2.5 times. At any other piston position the gas leakages will lay inside the multiple of 2.5 times.

2.The wear of the contact surface between the cylinder and piston where there is constant or periodical contact. Before a piston jams or destructs on its side surface, the wearing factors give a negative efforts on compressor operation primarily due to leakage increasing through gap seal.

3. An analog to the cause 2 effect on cylinder and piston wearing influence is by gas erosion caused by a large gas velocity and primarily quick gas pressure oscillation inside the gap.

All three types of wear can be controlled using helium leakage through the gap seal with a constant pressure difference along the gap.

In order to perform an accelerated test it is necessary predict leakage over a period of time (for example, after 10 years operation) using the results of real-time testing for a

one-year period. Study of the effect of the above possible mechanisms of wear lead us to consider the gas erosion mechanism as a significant factor. However, the dependence of wearing rate on the method of wearing is nonlinear. As we approach critical values, i.e. gap gas velocity, cycle frequency, frequency and level of helium pressure oscillation, etc., or quickly changing gas flow direction inside gap than it appears there is a sharp increase in the rate of surface wear or possibly even surface destruction (this is almost impossible to predict). The pulse tube problem is a result of a large external velocity gas being rapidly compressed within the volume of the tube and its compression energy being transported as heat through the tube wall material.

For the gas erosion mechanism the critical step is the presence of cracks on the surface of the cylinder and piston. Due to this the piston and cylinder must be produced from materials which are not subject to the crack development process. In addition to taking into account the contact and friction between piston and cylinder, the piston must be covered by a material with a low coefficient of friction and large a resistance to wearing. SRDB has significant experience in using diamond-like materials for such a cover application which helps us characterize the effects of the manufacturer's system.

We will consider the compressor designer's piston material selection and its sensitivity to crack development and its tendency for or resistance to rapid change of wearing intensity and piston destruction by the gas erosion mechanism. The piston wear rate by contact friction mechanism has relatively simple characteristics. Therefore, gas leakage increasing due to membranes suspension deformation is not more then a factor of 2.5 times. It is necessary to note, also, that wear during high frequency testing differs from low frequency wear because of the presence of more high moving rates and hence higher heating of the friction zone. The obvious result is that there is more wear of the piston during high frequency testing.

Our methodology of accelerated tribotechnical testing is based on the following:

- For a properly designed friction unit it must to be capable of accommodating changes in the operating parameters (velocity of sliding, normal loading, temperature, etc.) with limits which encompass the anticipated regime of "stationary friction".

- Regimes of "stationary friction" are characterized by the constant coefficient of friction and wear characteristics during total operational lifetime of the friction sensitive unit, i.e. piston and cylinder.
- The operational duration ("friction durability") of the friction unit is the ratio of the operating time or number of cycles which relate to the elements of the friction unit such as either the piston or cylinder or gas, to the time to when damage occurs. We monitor the following control operating parameters (gap dimension, value of leakage, friction losses) over given limits.
- In order to predict friction durability of the friction sensitive element we use a short duration real-time test process as an experimental-calculation method. It is based on building a diagram-of-dependence of the wear intensity over the stationary operating range from the operating time (number of wearing cycles). Our friction unit is built using results of series of short-time testing with its next linear approximation to a given time (number of operation cycles).
- For gas compressors with gap seals within the piston-cylinder it is necessary to do a parallel test of not less than two compressors operating aT standard and high frequencies. During these preliminary tests the tests are periodically stopped for the measurement of gas leakage.

Taking all of the above into account, we propose the following methodology for accelerated testing of leakage of gap sealing. For one (1) year test two compressors per the following:

•1 with standard operating frequency, and

•2 with high frequency operation.

•Testing of both compressors is periodically (aT three month intervals) terminated; stopping for measurement of leakage through the gap seal.

•For 1.5 to 2 years dynamic leakage change is measured.

•For high frequency compressors this test represents 6 to 10 years of operation. In order to accomplish this the frequency must be increased between 4 and 6 times. If during 1/6 to 1/10 of the test period the leakage of the high frequency compressor is equal or exceeds the

leakage of the standard frequency tested compressor it is possible to predict the leakage over 6 to 10 years and terminate testing.

3.2.2. A description of the principles of stand operation.

The schematic of the stand for checking the wear of Stirling cryocooler compressor gap sealing is submitted for the fig. 2.1. The stand equipment is illustrated in fig. 3.2.1. The functional stand design is shown in fig. 3.2.2. The pneumatic-hydraulic design is shown in fig. 3.2.3.

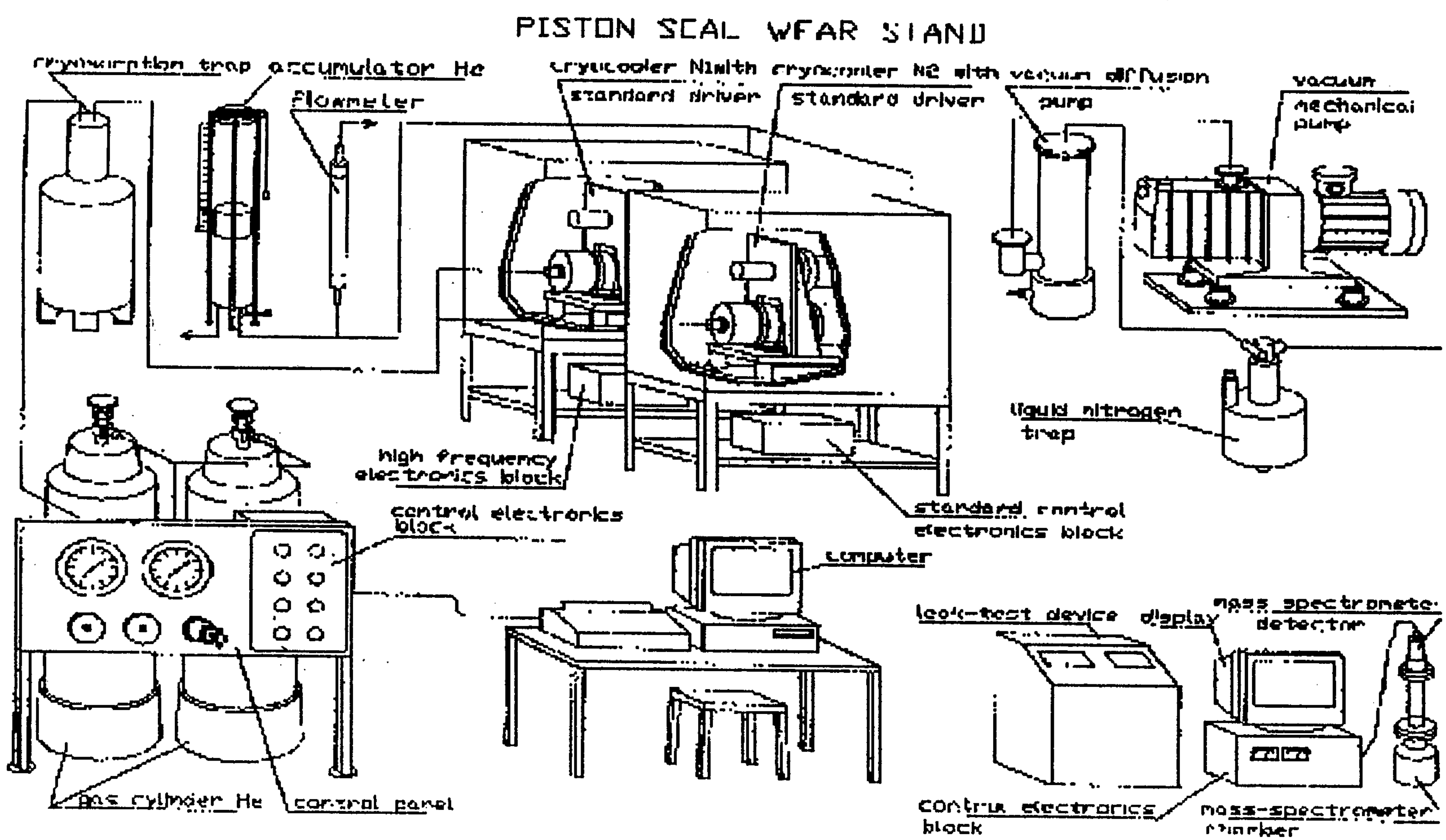
PISTON SCAL WFAR STAND
accumulator He
flowmeter
standard driver
standard driver
pump
vacuum
mechanical
pump
liquid nitrogen
trap
high frequency
electronics block
control electronics
block
standard control
electronics block
computer
leak-test device
display
detector
gas cylinder He
control panel
electronics
block
mass-spectrometer

Figure 3.2.1

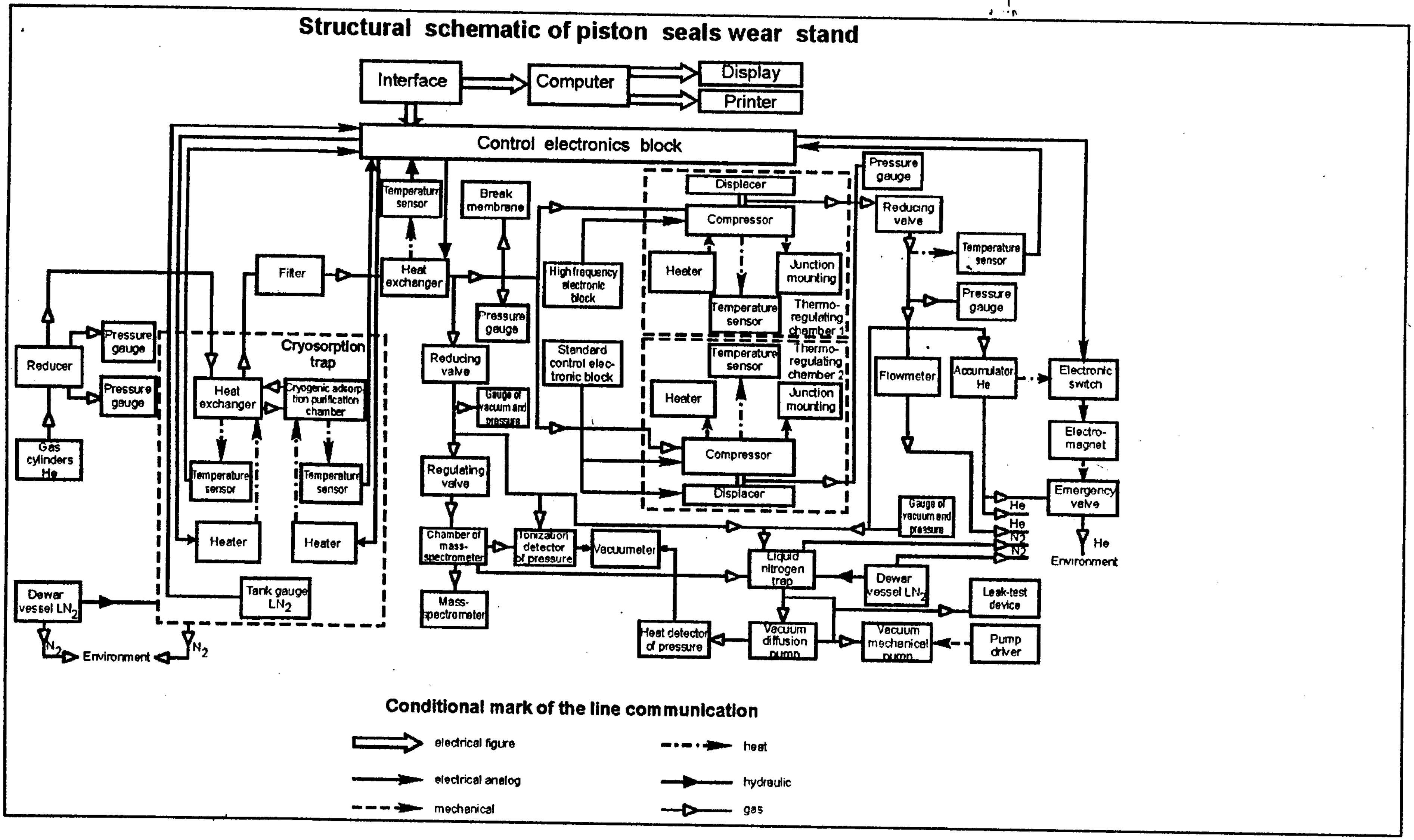
Structural schematic of piston seals wear stand
Interface
Computer
Display
Printer
Control electronics block
Temperature sensor
Break membrane
Pressure gauge
Reducing valve
Displacer
Compressor
Heater
Junction mounting
Temperature sensor
Thermo-regulating chamber 1
Thermo-regulating chamber 2
High frequency electronic block
Standard control electronic block
Filter
Heat exchanger
Cryosorption trap
Cryogenic adsorption purification chamber
Reducer
Gas cylinders He
Dewar vessel LN2
Tank gauge LN2
Environment
N2
Regulating valve
Gauge of vacuum and pressure
Chamber of mass-spectrometer
Mass-spectrometer
Ionization detector of pressure
Vacuumeter
Flowmeter
Accumulator He
Electronic switch
Electro-magnet
Emergency valve
He
N2
Liquid nitrogen trap
Vacuum diffusion pump
Vacuum mechanical pump
Heat detector of pressure
Leak-test device
Pump driver
Conditional mark of the line communication
electrical figure
electrical analog
mechanical
heat
hydraulic
gas

Figure 3.2.2

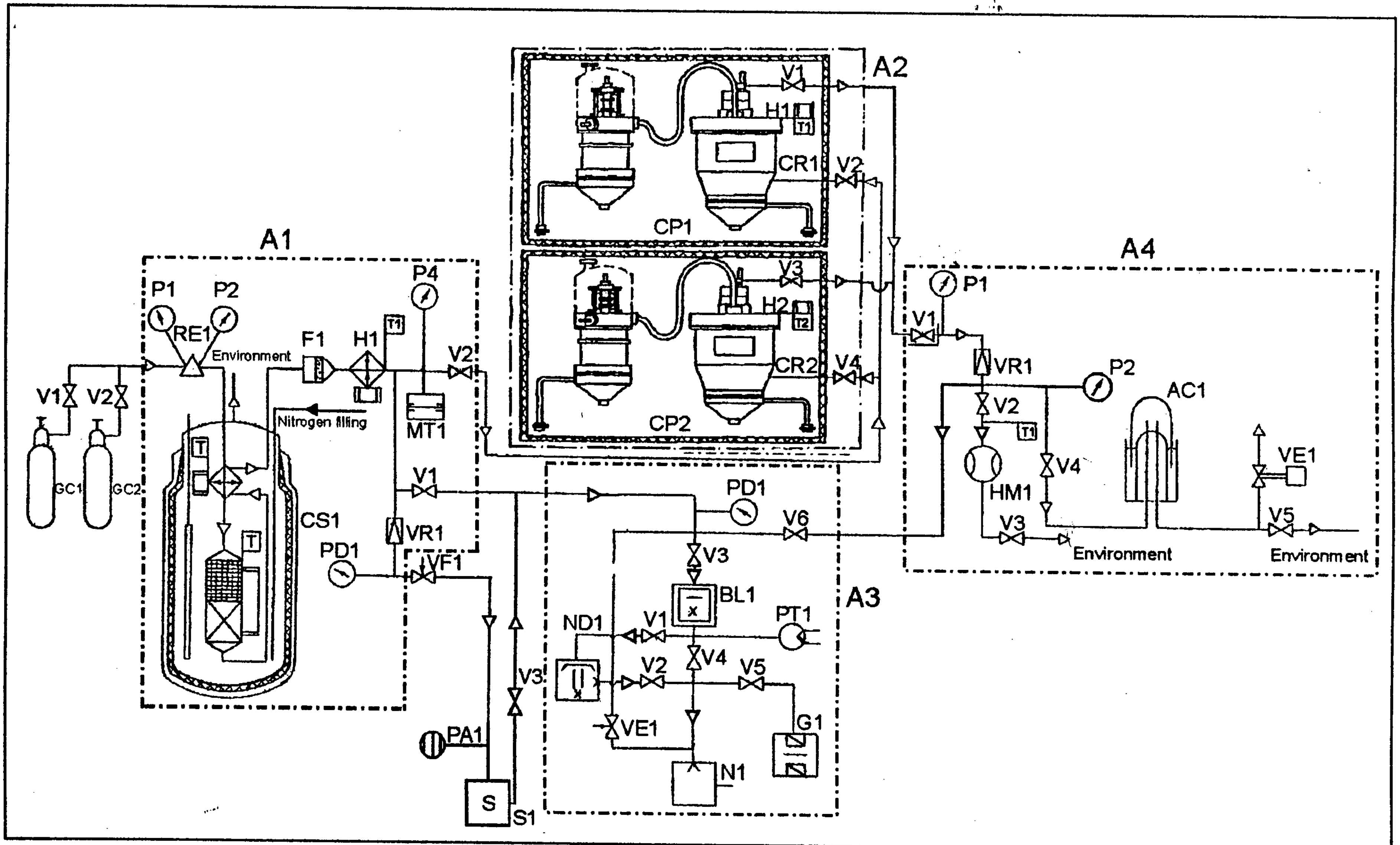

Figure 3.2.3 Principle Design of Stand for Checking Wearing of Stirling Cryocooler Compressor Gap Seal

The stand includes the following main parts:

- Two thermal regulating chambers with the inlet-outlet system of helium;
- A high pressure gas cylinder vessel for pure helium;
- A pure helium filling block with cryosorption trap;
- A mass-spectrometer system for checking the admixture quality in helium;
- A helium flow meter block;
- An oil-free vacuum pump block;
- A stand control block.

The stand allows us to conduct a parallel test of two Stirling cryocoolers. Exercised cryocoolers - one with the increased operating frequency (cryocooler 2), and the other with the nominal frequency (cryocooler 1) are placed in their respective thermal regulator chambers (CP2 and CP1 accordingly). Cryocooler electronic controller standard block for cryocooler 1 and high frequency block (developed by SR&DB) for cryocooler 2, are placed outside of the thermal regulating chambers. Here and on all the following equipment the equipment elements are marked per the principle pneumo-hydraulic plan (fig. 3.2.3).

The thermal regulating chambers must satisfy several problems:

- Provide attach points for the tested cryocoolers;
- Maintain a stable with outside temperature of 30 to 40° C during test;
- Ensuring a cryocooler temperature with a range of -20 to +40° C under accelerated testing of cryocooler 2 compressor and normal work of cryocooler 1.

Below is a description of the various required components:

1.In order to ensure a given temperature the test chamber is provided with electric-heaters Í1 and Í2 as well as sensors for checking the temperature aT the attach points of the cryocoolers 1 and 2.

2.Armature valves V1 to V4 are required for connecting the measurement equipment and for performing a fluid path for either of the two tested compressors. The inlet-outlet helium system for the compressors is built to accommodate the pressure difference applied to the compressors piston from the carter side. This is incorporated to stop any unacceptable deformation of the lavaliere of the piston suspension under the

displacement of the carter side (if in the design of the compressors this is absent a piston moving limiter is incorporated).

3.We assume that standard thermal chambers, slightly modified to accommodate the inlet-outlet helium system will be provided by the US Customer. We plan on specifying the inlet-outlet system but it can be cost effectively be purchased in the US since all parts are off-the-shelf.

4.High pressure standard gas cylinders, GC1 and GC2, for pure helium are designed to store adequate helium required for a 2-year test cycle. This is the maximum test time required to the wear of the gap seal of two Stirling cryocooler compressors. Gas cylinders for pure helium and their armature should be included in standard equipment provided by the US Customer.

5.The pure helium filling block, A1, ensures clean helium is put into the test cryocoolers. The provided reducer, PE, pressure gauges, P1, P2, P4, and temperature heat exchanger - heater H1 and temperature sensor - T1 are included in the filling block. The required high level of helium purification from mechanical particles and gaseous admixtures, which can form cryodeposit in the displacer block, is ensured by means of a cryo-adsorption trap, CS, and a mechanical filter, F1.

6.A stop valve, V2, and a reduction valve, VR1, along with the manometer, P3, are used to connect the vacuum system and check the quality of helium purification.

7.The safety membrane, MT1, protects the cryocoolers from any excessive operating pressure. The block for helium preparation, along with the helium source (gas cylinders), ensures the gaseous helium entering the cryocoolers will be aT the required pressure under the given temperature after the completion of following cycle of compressors sealing gap testing.

8.The block for helium preparation will be designed and produced in SR&DB. The mass spectrometer, S1, is intended for periodic quality checking gaseous helium to assure no contamination from admixtures which can form cryodeposits in the displacer of cryocooler. The ionization manometer, PÀ1, and throttle-valve, VF1, are included in the mass spectrometer kit for the vacuum check in the camera of mass spectrometer. The throttle valve, VF1, provides gaseous He aT atmospheric pressure during the filling

displacement of the carter side (if in the design of the compressors this is absent a piston moving limiter is incorporated).

3.We assume that standard thermal chambers, slightly modified to accommodate the inlet-outlet helium system will be provided by the US Customer. We plan on specifying the inlet-outlet system but it can be cost effectively be purchased in the US since all parts are off-the-shelf.

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8.The block for helium preparation will be designed and produced in SR&DB. The mass spectrometer, S1, is intended for periodic quality checking gaseous helium to assure no contamination from admixtures which can form cryodeposits in the displacer of cryocooler. The ionization manometer, PÀ1, and throttle-valve, VF1, are included in the mass spectrometer kit for the vacuum check in the camera of mass spectrometer. The throttle valve, VF1, provides gaseous He aT atmospheric pressure during the filling

process of the vacuum chamber through mass spectrometer from the pipeline. The mass spectrometer camera evacuation to the operating pressure is accomplished by means of the oil-free vacuum block, A3. We anticipate that in the stand standard mass spectrometer supplied with an auxiliary and vacuum equipment will be provided by the US Customer.

9.The helium flow meter block includes a block of measurement devices for measurement of helium flow rate, A4, temperature sensors, T1 and T2, located within cameras, CP1 and CP2, respectively and manometer, P4, located in the helium preparation block, A1. The block for flow rate measurement allows us to check the helium flow through the compressor gap seal in quantity measurements, liters per hour, and volume, cubic meters per hour. In order to perform these functions it is supplied with removable flow meters, HM, for the measurement of greater helium rate and a known volume accumulator, AC, for the measurement of small rates of helium by the method accumulation. Placing a given helium pressure on cryocoolers output is ensured by the means of a regulating valve, VR1, and valves, V2-V4, but its measurement is performed by means of the manometer, P1. By means of an electromagnetic valve, VE1, and the accumulator, AC, built-in end breaker the system is protected from the overflow. The system for checking the helium flow parameters will be designed and produced by SR&DB.

10.The vacuum block, A4, is intended for solving the following problems:

- Prepare the stand for operation (pump out and hermetically check);
- Ensure the mass spectrometer continues to periodically check the admixtures in the helium;
- Provides adsorbent regeneration in the block for helium purification;
- Check of the Stirling cryocooler hermetic state after its start-up.

The block has the following composition:

- The mechanical vacuum pump, N1, with the evacuation rate not less than 3 l/s;
- The diffusion pump, ND1, with a evacuation rate (in interval of pressures from 10^{-5} to 2×10^{-4} Hg mm) of not less than 100 l/s;
- A trap for liquid nitrogen, BL1;

- Vacuum pipelines and an armature for connecting the pumps and traps having diameters not less than 30 mm;
- Vacuum meter, PT1;
- Helium leak-test device.

It is expected that the vacuum block will be assembled from standard equipment and delivered by the USA side.

11. The controller block of the stand is intended for solving the following problems:

- autocontrol of the electric heaters of the cryo adsorption block for helium purification, CS, heater, H1, thermal regulating chambers and CP1 and CP2 by means of corresponding sensors of temperature;
- Failures identification (overflow of accumulator, AC, breakup safety membrane, MT1, liquid nitrogen level reduction in the trap or purification block to the minimum limit, rejection of the cryocoolers controller block, etc.);
- Temperature measurement;
- Storing results of all parameters measurements (rates, pressures, temperatures, features of cryocoolers, etc.);
- Accumulating and organizing the databases of test results;
- Generation and distribution of reports.

All such operations as control of the stand stop valves, tuning of the reducer and reducing valves, cut-in-unhooking of the vacuum pumps, of leak-test device and of mass-spectrometer and other single operations are performed manually. The controller block of the stand is developed, produced and delivered by SR&DB.

During all periodic testing in the nominal regime (heat lift capability, power consumption, vibration disturbance etc.) of cryocoolers 1 and 2 standard US equipment (not shown on fig. 3.2.1 - 3.2.3).

The main technical characteristics of the stand are given in table 3.2.1.

Table 3:2.1

Row #	Technical parameter	Parameter value
1	Number of exercises simultaneously cryocoolers	2
2	Stand working medium	Pure helium
3	Gaseous admixture quality in helium after purification, not more than, g/m^3	$1 \cdot 10^{-4}$
4	Temperature range of control temperature system, °C	+60...-50
5	Pressure range of control pressure system, MPa	0.1-15.0
6	Mass-spectrometer analyzed mass range	2- 600
7	Helium pressure inlet compressor, MPa	0.3- 1. 5
8	Helium pressure difference on compressor gap sealing, MPa	0.1- 1.0
9	Helium temperature inlet compressors, ^{0} C	30- 40
10	Helium flow rate range, l/h	1 - 5000
11	Duration of one blowing through gap sealing, min.	1 - 10
12	Total pure helium reserve in gas cylinders aT pressure 15 MPa, not less, m^3	6
13	Evacuation rate of mechanical vacuum pump, not less, l/s	3
14	Evacuation rate of diffusion pump (aT pressure $1 \cdot 10^{-5}$- $2 \cdot 10^{-4}$ Hg mm) , not less, l/s	100

3.2.3. Calculation results of the primary stand parameters.

The primary purpose for the stand calculation is the determination of the expected range of the helium flow rate. It depends on the following conditions to make the correct choice of measurement facilities and other equipment. In addition, we must evaluate the helium reserve in the stand, required for a full test cycle.

Helium leakage through gap sealing depends on many parameters: geometric gap amounts, pressures difference in compressor cavities; gas temperatures; relative

positions of the piston and cylinder a forming gap; mode of gas current the along gap walls and the velocities of relative piston displacement.

We have chosen the following conditions for test. The piston is still and located in an extreme position corresponding with a minimum cavity volume of compression. The helium pressure aT the input of gap sealing (aside of the cryocooler carter) is installed aT the level corresponding to the average pressure of leading-in: for the BAe cryocooler - near 1.4 MPa. Pressure difference through gap sealing is chosen small enough to provide an absence of the helium critical current and to get measurements sufficiently accurate to determine the given pressures by the stand facilities (manometers, reduction valves, etc.). The temperature of the helium provided to the cryocooler and the temperature of its attach points in the thermal regulating chambers are alike aT a level of 30-40°C.

In Walker G. work of "Cryocoolers" [part 2: Applications, chapter 9: Same aspects of design. Close tolerance seals., Plenum Press, New York, 1983] flow rate of gas through cylindrical gap sealing in cryocoolers it is recommended that the formula be define as:

$$Q = A\frac{d\delta^3 \Delta P}{L}; \tag{3.2.1}$$

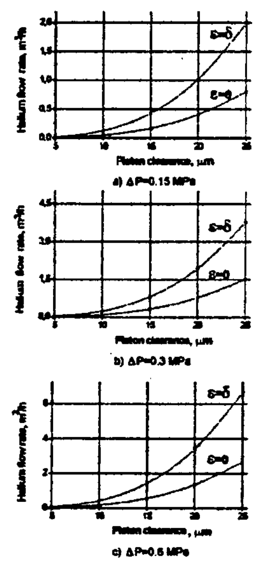
where A - the gap constant depending on the gap geometry and gas characteristics; d, δ, L - accordingly the piston diameter, width and length of gap; ΔP - a gap pressure difference.

In actual conditions of the compressor operation, the piston and cylinder axis can be displaced parallel on certain distance one from one another due to inaccuracy of fabrication or radial deformation membrane suspensions. For such events, helium flow rate through gap sealing (see reference book "Sealing and sealing technology", reductor A.I. Golubov, Ì., Machine building, 1986) is defined by the formula:

$$Q = \frac{\pi \cdot d \cdot \delta^3 \cdot \rho \cdot \Delta P}{12\mu L}\left[1 + 1.5\left(\frac{\varepsilon}{\delta}\right)^2\right]; \tag{3.2.2}$$

where ε - the value of displacing the piston and cylinder axes; μ and ρ- accordingly viscosity and density of a helium under the average pressure in gap.

Fig. 3.2.4. Helium flow rate across a gap sealing.



From the above results it follows that the stand measurement facility of helium flowing through gap sealing of the BAe compressor must allow us to measure the helium flow rate within limits from several liters per hour to one thousand liters per hour. In order to ensure sufficient accuracy of stand measurements we have provided two methods for the determination of the helium flow rate; a removable rotor flow meter and a helium volume accumulator.

Pure helium store in tanks of the stand must suffice for the whole cryocooler test cycle. This us to avoid an inevitable stand contamination during the connecting-disconnecting of stand pipelines.

The test is planned for two cryocoolers. The numbers of tests for each cryocooler will not exceed 8 times in repetitions of three phases. This means that even under the average helium flow rate of 500 l/h and a duration of each blowing through to 10 minutes, the total helium flow consumption just on blowing through the cryocoolers will not exceed $4m^3$.

Taking into account helium usage for leading-in the cryocoolers after each cycle of test and residual helium in gas cylinders and inevitable technological losses during

stand preparation with al pure helium reserve in the gas cylinders of the stand it will be less than 6 m^3 used.

Practically (with twice the reserve) the helium amount is kept in two standard gas cylinders with a capacity of 40 l each aT the pressure 15 MPa.

3.2.4. Primary requirements of the standard USA equipment.

3.2.4.1. GC1 and GC2 gas cylinders.

GC1 and GC2 cylinders with stop valves intended for keeping super pure gaseous helium aT pressures before 15 MPa.

Volume for each cylinders must form not less than 40 l, but the working pressure will be Pw = 15 MPa.

3.2.4.2. Mass spectrometer, S1.

Mass spectrometer, S1, is intended to periodically check the gaseous helium purity for admixtures which can form cryodeposits in the displacer block of the Stirling cryocooler.

Range of analyzing masses 2 - 600 masses.

Throttle-valve, VF1, and ionization manometer, PÀ1, must enter In the kit of the mass spectrometer for checking the vacuum in the camera of the mass spectrometer with the range of pressures from $2.4*10^{-5}$ to 0.13 Pa.

Throttle-valve VF1 is a valve for fine regulation and is used for gaseous He filling aT atmospheric pressure in the camera of mass-spectrometer from the pipe line.

3.2.4.3. Stop Valves, V1, V2.

Valves are intended for:

a) performing through cryocooler blowing;
b) sealing an internal cryocooler cavity after the next test cycle on checking of gap sealing wearing.

Working pressure of the valves is not less than 4 MPa; the conditional diameter is 2 - 4 mm; the helium leakage through each of closed valves aT the pressure of 2 MPa is not more $1*10^{-6}$ m^3Pa/s.

3.2.4.4. Thermal regulating chamber, CP1 and CP2.

Thermal regulating chambers, CP1 and CP2, are intended for:

a) the exercised cryocoolers fastening; b) for the maintenance of stable temperature of cryocoolers under helium blowing through aT temperature 30-40° C;

b) ensuring a temperature of cryocoolers within the range of from -40 to +40° C.

In order to ensure the conditions for blowing through are proper the chamber must be provided with electric heater Í1 and Í2, as well as sensors for checking a temperature of fastening places of cryocoolers 1 and 2.

3.2.4.5. Vacuum block À3.

Vacuum block is intended for:

a) stand preparation and to verify operability (evacuation and permeability check);

b) ensuring a working of the mass-spectrometer under periodic checking of the admixtures in the helium;

c) for adsorbent regeneration in the block for helium purification.

The evacuation block must include the following:

- mechanical vacuum pump with evacuation rate of not less than 3 l/s;
- diffusion pump, with evacuation rate (in the interval of pressures from 10^{-5} to $2*10^{-4}$ Hg mm) forms 100-150 l/s;
- trap for liquid nitrogen;
- vacuum armature for connecting the pumps and traps with a diameter of not less 30 mm;
- helium leak-test device.

3.2.4.6. Supply tank for liquid nitrogen.

Liquid nitrogen is required for filling the cryogenic filter-adsorber of the purification block and for the nitrogen trap of vacuum block.

Volume of tank must be chosen to have a total capacity for supplying not more than 15 l per a day.

3.2.4.7. Standard electronic controllers for BAe cryocoolers.

Standard electronic controllers must ensure a normal state of the working cryocooler 1 during and periodic checking of the functionality of cryocooler 2 after the high frequency testing a drive.

3.2.4.8. Standard equipment for performing test.

During cryocoolers 1 and 2 periodically testing requires use of miscellaneous standard USA equipment for cryocoolers performing test on nominal regime.

3.3 THE STAND FOR THE ACCELERATED TESTS OF THE COMPRESSOR BODY ON GAS TIGHTNESS UNDER THERMAL CYCLING

3.3.1. Substantiation of the accelerated tests technique.

Onboard cryocoolers, placed in conditions of orbital space flight are subjected just as the total space vehicle to regular cyclic changes of temperature of its body with a period equal to a cycle time of a space vehicle of one orbit of the Earth (typical cycle time is about 90 minutes). Such changes of temperature are connected to the periodic transition of the satellite from a shadow site of an orbit to solar one and vice versa. In a general case, depending on radius of an orbit and orbital space flight, on its flight-path angle to the Earth's equatorial plane, the change of temperature of the object can involved a range of -150°C to +100°C. According to the technical requirements of space cryocoolers manufacturers of Stirling type such as British Aerospace; Space Systems Limited or PSC type, Protoflight Spacecraft Cryocooler, the given devices should have a useable life with periodic changes of temperature from -60°C up to + 75°C during transportation, storage, pre-launch check, launch and orbit modes.

Cryocoolers loss of serviceability under influence of the specified temperature differentials is directly related to the loss of compressor or expander body gas-tightness. The body of the compressor and expander represents a tight cylindrical thin-walled vessel filled with gaseous helium under pressure approximately 15 bar. The presence of internal pressure results in the material of the compressor body to be in a constant static

two-axes stressed state. With this background, the cyclic changes of the cryocooler body temperature results in the occurrence of cyclic temperature stresses in body walls caused by temperature gradients between external and internal surfaces of the body. During a 10-year stay on orbit the cryocooler body experiences about $6*10^4$ thermal cycles and the same amount of cycles of sign-variable thermal stresses. The amplitude of these stresses is defined, on the one hand, by a combination of the body's material thermal & mechanical properties (thermal conductivity λ, coefficient of linear thermal expansion α, Young's module E), and rate of the material temperature change $\Delta\dot{T}$. The smaller the value of λ the larger the value of α, E and $\Delta\dot{T}$, i.e., the greater is the amplitude of cyclic thermal stresses. These stresses also create a two-axes stressed state. Thus, the compressor and expander bodies are subjected to two-axes static stress due to internal pressure imparted by the helium in combination with two-axes cyclic loading due to thermal stresses. These products operations can be typified by the two-axes asymmetric fatigue cycle mode.

During long-duration cryocooler operation there is a high probability of an accumulation of damages in the body material. It is caused by the nucleation and propagation of fatigue microcracks. The result is unacceptable gas permeability within the body and a loss of its gas tightness.

The described situation requires the performance of tests of the compressor and expander bodies for gas tightness under full-scale conditions. However, taking into account the requirement of a ten-year cryocooler service life, the realization of such tests in real-time makes no sense. Therefore, there is a requirement for the development of methods and special equipment for creation of an accelerated test methodology with conditions emulating the full-scale regime.

The primary requirement for the methodology of the accelerated test process is the preservation of identity of the physical aging mechanism of the tested element(s) during accelerated and full-scale conditions. The element must experience all of the factors which cause aging. Only if this requirement can be met can the accelerated test be meaningful.

It is necessary to note, that in our case there has been placed an internal compensation. This has been described previously in this report under the complex stress state, i.e. gas static and the cyclic thermal stresses are counterbalanced not in grips, but in the wall of the compressor (expander) body. If the gas static component of the stresses acts over the total cross section of the wall, the maximum thermal component of stress is realized in superficial layers of the material. Therefore, the possibility for modeling the given situation through mechanical tests is not obvious.

The accelerated test maximum simulation of the real situation can be realized by the thermal cycling of the compressor (expander) body filled with gaseous helium under atmospheric pressure. This is done aT the expense of simultaneous change (in comparison with full-scale testing) of one or two of two thermal cycles parameters such as:

i - increase of the temperature amplitude of a " heating - cooling " cycle;

ii - reduction of the temporal period of a " heating - cooling " cycle.

Using of the first factor (increase of the temperature amplitude of a cycle) is rather attractive. However, it is unacceptable because of absence of reliable criteria for similarity with the real time state. If there were reliable criteria it would allow us to receive adequate result with a smaller amount of thermal cycles.

The reduction of a temporal period of thermal cycles with preservation of the regular values of their amplitude is related to the increase of rate of heating and cooling of the body. This will result, on the one hand, in an increase of the thermal cyclic stress components of amplitude, and, on the other hand, to the reduction in time-of-stay of the material of the body under action of maximum thermal stresses. Since the range of operating temperatures of the cryocooler body is much lower than the temperature with which the processes of active creep in metal constructional materials develop [1], the influence of the second (temporal) factor is insignificant. At the same time, the increase of amplitude of cyclic the thermal stresses connected with increased $\Delta \dot{T}$, should not be removed from the following restriction frames.

The rate of heating and cooling during an accelerated cycle should be reduced so that the rising variable thermal stress component, (taking into account

the background gas statistics of stresses) does not cause essential structural changes in the material (for example, phase transformations). The stress component will not exceed the materials fatigue limit, σ-1 on the basis of $5 \cdot 10^6$ cycles.

The first requirement is satisfied with maintenance of structural stability in the given temperature interval material of the body. The second requirement is met by establishing a base number of cycles exceeding by two orders the number of full-scale thermal cycles for ten years of orbital flight. Therefore the chosen fatigue limit, σ-1 will contain a sufficient safety factor guaranteeing that fatigue fracture of the body with accelerated thermal cycling tests will not occur.

In the following section the results of our theoretical analysis and experimental check of acceptable (from the point of view of the stated restrictions) regimes of accelerated thermal cycling tests will be given.

At the conclusion of this section it will be seen that by virtue of similarity of design, materials and operational conditions of the compressor and expander bodies, and further with the test stand design, we shall realization the viability of accelerated tests for gas-tightness of the compressor body, only. Thus, by maintaining the criteria for gas-tightness of the body after action of the given number thermal cycles will be absence of outflow helium, determined with the help high-sensitivity helium leak-meter.

3.3.2. Results of estimation and experimental validation of the stand characteristics

We shall initially consider the stressed state of the compressor body without taking into account the rising thermal pressure. The model represents the thin-walled cylindrical vessel, closed from both sides, which is under internal pressure P. The material body in this case is in planar tension stressed state with the primary stresses, σ1 (tangential) and σ2 (axial).

$$\sigma_1 = \frac{\pi D}{2t} \cdot P \quad \text{and} \quad \sigma_2 = \frac{\pi D}{4t} \cdot P \quad , \qquad (1)$$

where D - diameter of the vessel, t - thickness of a vessel walls.

Assume that the diameter of the compressor body ≈110 mm, the internal pressure of gaseous helium is ≈15 bar and chosen thickness of a wall is ≈ 3 mm. Then the primary stresses are by equations 1 - $\sigma 1= 8.6\ kg/mm^2$; $\sigma 2= 4.3\ kg/mm^2$.

As a result of cyclic external heating and the cooling of the cryocooler body the variable radial thermal flow rises. This causes, in a general case, the material of the body to have a field of variable thermal tension or compression stress with the maximum values on the material's surfaces. The field of thermal stresses is characterized by the primary tension or compression stresses, $\sigma 1\text{Ò}$ (tangential) and $\sigma 2\text{Ò}$ (axial), with the relation $\sigma 1\text{Ò}/\sigma 2\text{Ò} \approx 1$.

Thus, the field of the stresses causing a planar tension stressed state with constant primary stresses, $\sigma 1$ and $\sigma 2$, is combined with the two-axes field of thermal stresses of tension or compression with variable values and the primary stresses are $\sigma 1\text{Ò}$ and $\sigma 2\text{Ò}$. The resulting field will be characterized by the stresses $\sigma 1P$ and $\sigma 2P$:

$$\sigma 1\text{Ð} = \sigma 1+\sigma 1\text{Ò}(\tau, T) \qquad \text{and} \qquad \sigma 2\text{Ð} = \sigma 2+\sigma 2\text{Ò}(\tau, T); \qquad (2)$$

where τ - time, T - temperature, $\sigma 1\text{Ò}$ è $\sigma 2\text{Ò}$. - amplitude value of thermal stresses.

Now we shall move from the flat stressed state (tension or compression) to the equivalent uniaxial stressed state (tension or compression), using the hypothesis of the maximum normal stresses [2]. The equivalent linear condition of stretching (compression) is characterized by the greatest main stresses $\sigma 1P = \sigma ecv$. We now deal with an asymmetric cycle which is described in parameters σm (average constant stress of a cycle) and σa (amplitude of alternative stress component of cycle). In this case $\sigma m=\sigma 1$, $\sigma a=\sigma 1T$ and $\sigma 1P = \sigma ecv = \sigma 1+\sigma 1\text{Ò}(\tau, T)$

In order to estimate the thermal and resultant stress values it is necessary to know mechanical, thermal & physical characteristics of the structural material from which the cryocooler body is made. From our data aluminum and titanium alloys are usually used as the body's structural materials. For an estimation let us choose the representative structural Al - based alloy with high-strength alloy 1460 (Al-Cu-Li), and representative structural Ti- based alloys - alloy ÂÒ5-1êò (Ti-Al-Sn). Mechanical and

thermal & physical properties of these alloys under normal temperature are submitted in table 3.3.1. [3, 4,5].

Table 3.3.1.

Properties of structural alloys

Alloy	Ultimate strength σu, MPa	Yellow stress σ0,2, MPa	The Young's module E, MPa	Fatigue limit σ-1, MPa	Coefficient of linear thermal. expansions α, 10^{-6} K^{-1}	Coefficient of thermal conductivity λ, W/m·K
1460	550	480	$7.1 \cdot 10^4$	120	22	140
ÂÒ5-1êò	835	800	$1.1 \cdot 10^5$	225	9	11.0

In the general case the thermal stresses, σÒ, arising on the body surface with a change of temperature of the environment, are described by expression:

$$\sigma_T = \frac{\alpha \cdot E \cdot \Delta T}{1-\mu} \cdot f \quad ; \qquad (3)$$

where α- coefficient of linear thermal expansion; E - module of normal elasticity; μ-Poisson`s ratio; ΔT - temperature difference between the external environment and initial temperature of a body; f - function describing the field of temperatures in the cross -section of the body with time under changing boundary conditions.

We shall now consider a case of the most rigid thermal loading (so-called thermal shock). This is when the rate of cooling or heating reaches such values that only a thin superficial layer of the body accepts the temperature of the external environment, whereas all other materials maintain the initial temperature through material thickness. In this case, thermal deformation of a superficial layer passes completely to mechanical deformation. The function, f , in the equation (3) equals 1.

We shall substitute in (3) the value of other parameters, and ΔT in this case corresponds to maximum required temperature range operating on orbit and makes

Tmax -Tmin = + 75° C - (-75°C)=150 ° C. By this estimation $\sigma 1T=\sigma a\approx 33$ kg/mm² for the Al-based alloy and $\sigma 1T=\sigma a\approx 21.5$ kg/mm² for Ti-based alloy.

The allowable peak of the thermal stress, $\sigma 1T$, should not exceed the limiting values of σaR, in accordance with the fatigue limit for the given basic number of cycles of an asymmetric cycle with the given constant average stresses $\sigma m=\sigma 1$. This value is determined either experimentally or from the ratio associating the fatigue limit under the symmetric cycle loading, σ-1, with the parameters of an asymmetric cycle. One can used for this case the ratio:

$$\sigma_{-1}^{2} = \sigma_{m}^{2} + \sigma_{m} \cdot \sigma_{aR} \tag{4}$$

Substituting in eq.(4) the values σ-1 (from table 3.3.1) and $\sigma m=\sigma 1$ (from eq(1)) we obtain for Al - based alloy - $\sigma aR\approx 8$ kg/mm² and for Ti - based alloy -$\sigma aR\approx 50$ kg/mm².

Thus, in case of pulse heating or the cooling (thermal chock) for the based Al-alloy the rising thermal stresses are because $\sigma 1T>\sigma aR$. As for Al - based alloy $\sigma max=\sigma m+\sigma 1T=8.6 + 33=41.6$ kg/mm², that is comparable to value of $\sigma 0,2$. For Ti-based alloy -$\sigma 1T<\sigma aR$, and $\sigma max=8,6+21,5=30,1$ kg/mm², $\sigma max<\sigma 0,2$. Therefore thermal shock with the accepted amplitude ΔT = 150° C do not present appreciable danger.

In actual conditions of heating and cooling temperatures, gradients through the cross-section of the body are significantly less than in case of hypothetical pulse heating because of non-zero material thermal conductivity and heat exchange with the external environment. Thus, the thermal deformation in a superficial layer only partially passes mechanically. In the formula (3), it is reflected that the function, f, becomes less than 1.

Estimation of thermal stress values under specific conditions of cooling and heating of a body of the simple form (for example plate or hollow cylinder) can be carried out by measuring gradients of temperature between outer and internal surfaces of these same bodies and by accepting distribution of temperatures through the thickness as linear. Then the peak value of thermal stresses in a case of hollow the thin-walled cylinder will be equal to:

$$\sigma_{aT} = \frac{\alpha \cdot E \cdot \Delta T}{(1-\mu)} \quad ; \qquad (5$$

where ΔT - maximum actual difference of temperature between the external and internal surfaces of the plate or hollow cylinder arising during their heating or cooling which corresponds to the maximum actual thermal stresses. For the definition of the actual ΔT, the estimated experiments were carried out.

On fig. 3.3.1. we provide experimentally, dependency of external and internal surface temperature of plates by thickness of 6 mm for alloys 1460 and ÂÒ5-1êò under thermal cycle $\pm 75°$ C cooling of one side of the plates was carried out in liquid nitrogen and heated by direct thermal radiation. The duration of the heating-cooling period did not exceed 3.5-4 minutes. Duration of the half-cycle "heating" or "cooling" are defined by time of alignment of temperature of the bottom and top surfaces. The maximum difference of temperature between surfaces of a plate ΔT_{max} makes $\approx 7°$ C for 1460 alloy and $\approx 60°$ C for ÂÒ5- alloy. It is reversible proportional to the thermal conductivity. Therefore, the correlation is actually observed.

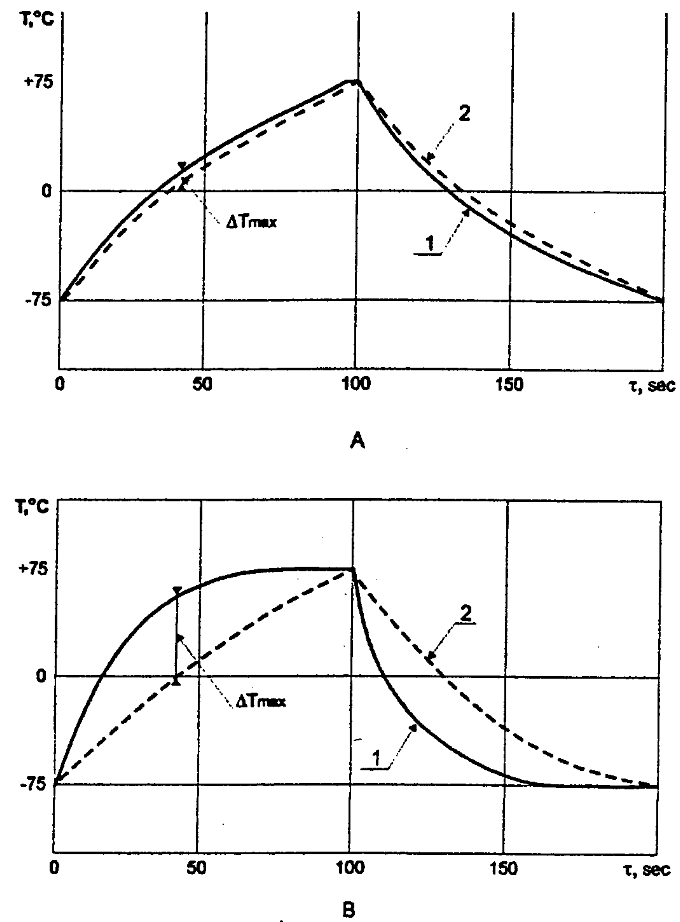


Figure 3.3.1.

After substitution of numerical values of the parameters in eq.(5) we obtain: σaT ≈1.5 kg/mm² for 1460 alloy and σaT ≈8.5 kg/mm² for ÂÒ5-1êò alloy. Thus, actual cycling is lower than fatigue limit of the material for the given test specification thermal stresses by a factor of approximately in 5.

Hence, it is possible to ascertain that the reduction of the temporal period of the thermal cycle τ from 90 min. up to 3.5-4 min. with a temperature amplitude of the cycle being ± 75° C. This does not represent a danger from the point of view of an opportunity for thermal fatigue as the material compresses under on thermal cycling accelerated tests.

THE STAND BLOCK DIAGRAM FOR ACCELERATED TESTS OF A COMPRESSOR BODY ON AIR-TIGHTNESS UNDER THERMOCYCLING

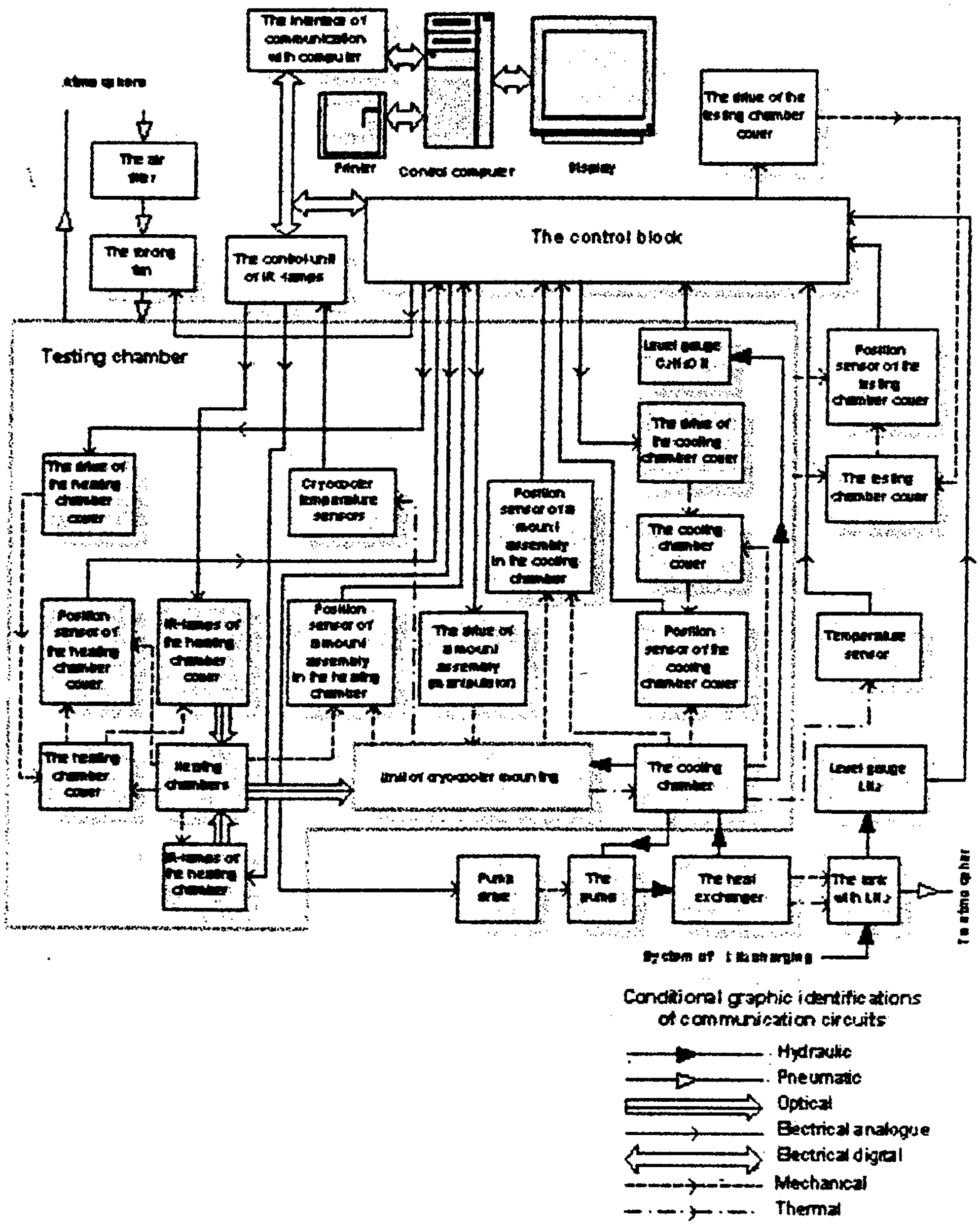


Figure 3.3.2

The stand components:

- test chamber (TC) ;
- chamber for heating (CH) ;
- chamber for cooling (CC) ;
- cryocooler mounting assembly with the manipulator (MA) ;
- system for liquid nitrogen (LN2) supply;
- system for ventilation (SV) ;
- electronic control system (EC);

The test chamber (TC) is constructed having a tight volume with the upper cover containing an electric drive, position gauge and ventilation system. Test for cryocooler body permeability to gas is performed with the upper cover opened the chamber by a He leakage meter. In the TC there are separate chambers for heating and cooling as well as a manipulator supplied by cryocooler mounting assembly.

The chamber for heating (CH) is constructed having a tight volume also supplied with an upper cover containing the electric drive and the position gauge. Inside the chamber and on an internal surface of a upper cover the infrared heating lamps are mounted.

Chamber for cooling (CC) is constructed as a thermally insulated vessel with a wide neck. It contains ethyl alcohol as cooling agent and a level meter. The CC has a tight upper cover supplied with a drive and the position gauge. In addition, for the purpose of refilling with the cooling agent, the CC is supplied with branch pipes. The cooling agent is input and output heat exchange for alcohol cooling in the LN2 vessel. If necessary, the level meter indications are made in the CC.

The cryocooler mounting assembly with the manipulator (MA) is constructed as a universal assembly. Its location is by a joint on the end of the manipulator. The mounting assembly contains a drive and three position gauges ensuring stop of the manipulator in CÍ, CC and in the average position.

The manipulator provides a constant spatial position of the assembly during all manipulations. Electrical communication lines for gauges of the temperature control of the test compressor body are located in a bar of the manipulator pass.

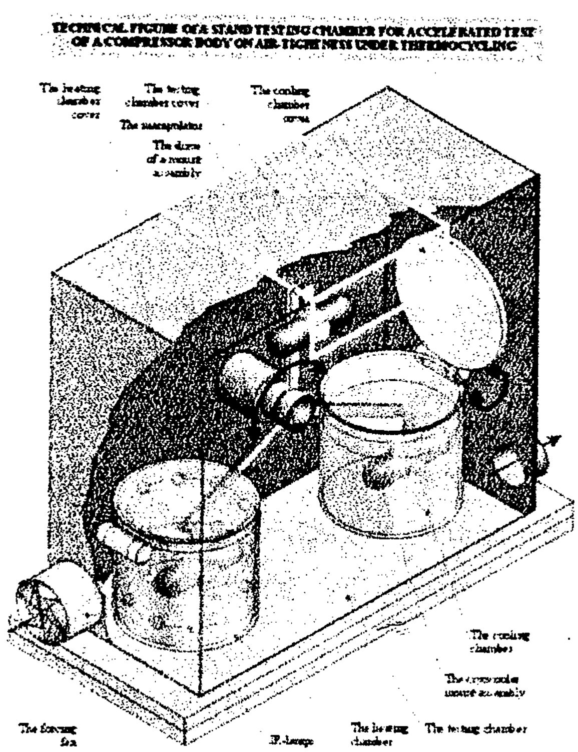
The system for LN2 supply includes the LN2 tank with large volume supplied by the level meter which indicates refilling of the LN2 tank is complete. At the bottom of the CC is mounted the heat exchanger.

The ventilation system (SV) pumps outside air through the TC by fan.

The control system (SC) consists of handling blocks which supply the operation of the CH infrared lamps, drives of the Cĺ and CC covers, and LN2 pump.

The handling blocks are connected to the control computer which is supplied with a display and printer.

Fig. 3.3.3



The principle of the stand operation consists of the following:

The test of compressor body is made by periodic heating and cooling. The heating is made in the CH by help of infrared lamps (IRL) with a given capacity. The lamp capacity regulation is provided by the handling block under the control of temperature sensors placed on the test compressor body. The cooling is performed in the CC constructed as a thermally insulated vessel filled by the cooling agent. The cooling agent, alcohol, is pumped through the heat exchanger which is located in the LN2 tank and then routed through the CC. The tested body is immersed in the cooling agent for the time required to cool to the desired temperature. The time for cooling is inputted by the handling block based on the indications of the temperature sensors.

The transportation of the tested cryocooler from the CH to CC and back is made by a special drive manipulator. On the end of the manipulator bar there is a mounting assembly where the tested cryocooler body is fixed. In order to make a body check for tightness the manipulator periodically stops in the average position. Here the special TC upper cover with a drive is located. After the cover is opened there is access to the cryocooler to check its tightness with a helium leak meter.

The chambers for heating and cooling are sealed with tight upper covers which open by their respective drives only for installation or removal. The MA is used for these operations.

The ventilation system is used for removal of the alcohol vapor from the TC which appears in it after cryocooler immersion in CH).

In CC the control of the cooling agent rate is stipulated by the level meter indications. With the level meter turned down the cooling agent is transferred. The cooling agent located in the CC is constantly pumped through heat exchanger located in the LN2 tank. The rate of transfer is adjusted by the control system under the temperature sensors located in the cooling agent.

The LN2 level in the tank also is measured by the level meter. If necessary the LN2 tank is refilled by a special refilling system. All manipulations of the upper covers of the chambers and the regulation of the cooling agent transfer are made by the handling block under the control of the position indicators, level sensors and temperature sensors.

The handling blocks work under the control of the computer which output the information on the display screen and printer.

The primary technical parameters of the stand are given in Table 3.3.2.

Table 3.3.2

Stand

for testing for gas tightness under thermal cycling

(the primary technical parameters)

Row #	The parameter name	Parameter value
1	Dimensions of test object, mm compressor displacer	 120 dia x200 length 78 dia x230 length
2	Mass of test object (compressor/ displacer), kg	3.24/ 1.09
3	Heating temperature (max), °C	+150
4	Cooling temperature (min.), °C	-196
5	Operation temperature cycle range, °C	± 75
6	Thermal cycle (heating- cooling) duration, minutes (depends on mass, geometry and material of testing object)	4-15
7	Transportation time from hot to cold bath, s	30
8	Object transportation method	by mechanical manipulator
9	Test regimes performance, including operations: heating-exposure-transportation-cooling-exposure-transportation-heating and so on	automatic
10	Acceleration coefficient (depends on operating conditions)	10 - 25
11	Power consumption (max) , kW	8

3.3.4. Primary Requirements for Standard USA Equipment

The stand for the accelerated tests of the compressor body on gas tightness should be completed with a high-sensitivity helium leak-meter and differential manometer of delivered by the US Customer.

4. CONCLUSION.

As a result of work carried out SRDB the basic approach for the accelerated tests long life space cryocoolers is established, the calculated and experimental substantiation of these approaches and methods of tests are completed and the general technical aspects of test stands as well as the content, arrangement and main technical characteristics of stands are developed for:

- the accelerated tests of an electrical linear drive of the compressor;
- a control of the wear of the compressor gap seal;
- the accelerated test of the compressor body for gas tightness under thermal cycling.

The stands allow us to realize the following scales of acceleration tests processes of the cryocooler compressor critical elements:

- for gap sealing wear, 5 to 10 and possibly more times;
- for checking the compressor body gas tightness under thermal cycling in, 10- 25 times;
- for work of the linear drive of the compressor, 4- 5 times;

The reduction of the last factor of acceleration to value 4- 5 (instead of previously predicted value 6- 10) is caused by Customer's exception of opportunity for entering the corrective modifications to the design of a linear drive. However we will be able to substantiate the basis for the accelerated test of the linear drive, regardless.

The preliminary study of the design of the stand for the accelerated tests cryocooler on failures connected with blocking of the expander by cryodeposit of products of outgassing of compressor elements has been initiated. For this stand the opportunity for acceleration of tests in 20 - 40 and more times is expected.

The work carried out to date is the first stage. Following will be detailed design, manufacture and delivery of the stands to AFRL. For effective flow to the next stages of detailed design and manufacturing of stands it is necessary to us to receive from the Customer the following additional data:

- presence of the piston position sensor in the test compressor;
- mass of the linear drive moving parts of the compressor and expander;

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- presence of the piston position sensor in the test compressor;
- mass of the linear drive moving parts of the compressor and expander;

- number of turns and wire length of the winding of the linear drive;
- magnetic field induction inside the gap of the drive magnetic system;
- the nominal size of gap between walls of the piston and cylinder;
- materials of the compressor and expander body,;
- materials and geometrical sizes of membrane springs of a drive suspension, amount of springs in packages.

It is necessary to coordinate questions on structure of the standard test equipment of American delivery and about elements of their connection with stands of SRDB ILTPE delivery. It is our plan to visit AFRL during the next two months in order to have a direct communication and understanding of the US shared efforts.

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REPORT DOCUMENTATION PAGE

Form Approved OMB No. 0704-0188

1. AGENCY USE ONLY (Leave blank)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	October 1999	Final Report

4. TITLE AND SUBTITLE

Accelerated Life-Time Testing of BAe-Stirling Cryocooler with Linear Drive, Part I, Fatigue Related Testing

5. FUNDING NUMBERS

F61775-99-WE047

6. AUTHOR(S)

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7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

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ILTP&E 47 Lenin Ave
Kharkov 310164
Ukraine

8. PERFORMING ORGANIZATION REPORT NUMBER

N/A

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

EOARD
PSC 802 BOX 14
FPO 09499-0200

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

SPC 99-4047

11. SUPPLEMENTARY NOTES

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Approved for public release; distribution is unlimited.

12b. DISTRIBUTION CODE

A

13. ABSTRACT (Maximum 200 words)

This report results from a contract tasking R&D Bureau of the Institute for Low Temperature Physics and Engineering as follows: The contractor will investigate unique approaches to accelerated testing of space cryogenic cooler technology.

14. SUBJECT TERMS

EOARD, Space Technology, Thermodynamics

15. NUMBER OF PAGES

54

16. PRICE CODE

N/A

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19, SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
UNCLASSIFIED	UNCLASSIFIED	UNCLASSIFIED	UL

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. 239-18
298-102